

**ENCIT2020-0753****ENERGY AND EXERGY ANALYSIS OF AN INTERNAL COMBUSTION ENGINE USING DIESEL RK SOFTWARE****Wilian Nascimento Silva**Graduate Program in Mechanical Engineering (PROPEM), School of Mining, Federal University of Ouro Preto, campus Morro do Cruzeiro, s/n, Bauxita, Ouro Preto – MG, 35400-000. [wilian.silva@aluno.ufop.edu.br](mailto:wilian.silva@aluno.ufop.edu.br)**Elisângela Martins Leal**Department of Mechanical Engineering, School of Mining, Federal University of Ouro Preto, campus Morro do Cruzeiro, s/n, Bauxita, Ouro Preto – MG, 35400-000. [elisangelamleal@ufop.edu.br](mailto:elisangelamleal@ufop.edu.br)

**Abstract.** *This paper presents the energy and exergy analysis of an internal combustion engine using the Diesel RK software. In the computational modeling, Diesel N2, Biodiesel and their blends is used as the fuel. The study initially consists of modeling the geometric factors of the engine, considering the speed of 1800 rpm and assuming that the load is constant since this type of engine is stationary and commonly used to drive generators and/or compressors. The results show the influence of the parameters on the behavior of the engine for each type of fuel applied, thus enabling a comparative study between the fuels used.*

**Keywords:** *Biodiesel, Diesel, blends, Diesel RK software, performance, engine.*

**1. INTRODUCTION**

Internal combustion engines are thermal machines for the production of brake power in several sectors, from mining to consumer goods, with fossil fuel being the main element in their supply. According to the IEA (2019), in 2018, 60.06% of the primary energy in the world came from fossil fuels, with a large part of this total being used in the transport sector. Such evidence brings focus on several issues, including economic and environmental issues.

Climate change emphasizes replacing fossil fuels with cleaner sources. Biofuel emerges as an alternative source to the use of fossil fuels, which besides being less polluting, have some attractive economic characteristics.

An interesting and considered fundamental feature is that biodiesel can be manufactured in any region of the planet, reducing the costs of exports. In the case of fossil fuels, some reserves in the world are distributed, and, according to Lora (2012), 65% of these reserves are in the Middle East.

In compression ignition engines, there is a need to apply diesel oil. However, this can be replaced by biodiesel, which is a fuel with similar physical properties and has an already renowned application in the market. However, the majority, not to mention the totality of biodiesel present on the market, have lower chemical properties than diesel oil, as does the lower heating value, making the demand for fossil fuel still high.

An analysis of the technical feasibility of using biodiesel compared to diesel oil, in most cases, encompasses only the use of the first law of thermodynamics. Thus, the precise analysis of the use of the energy provided by the fuel beyond the efficiency of its chemical reactions is not interesting. Therefore, it becomes interesting to apply the second law of thermodynamics and the concept of exergy for more careful analysis.

In this context, this paper aims to present the energy and exergy analysis of an engine operating with diesel oil, biodiesel, and their mixtures to demonstrate possible applicability of these fuels in internal combustion engines using the Diesel RK software.

## 2. METHODOLOGY

### 2.1 Geometric characteristics of the engine

The engine used in the study was the DC13 072A model from Scania, which originally operates with Diesel oil. Its geometric properties and operational characteristics are described in Table 1.

Table 1 – Geometry and Operational Characteristics of the engine Scania DC13 072A.

Engine description – geometry properties				
Number of cylinders	6 in line			
Working principle	4 Stroke			
Firing order	1-5-3-6-2-4			
Displacement	12,7 Liters			
Bore × stroke	130 × 160 mm			
Compression ratio	16.3			
Piston speed at 1500 rpm	8 m/s			
Piston speed at 1800 rpm	9.6 m/s			
Engine description – operational characteristics				
Parameters	Engine speed (1500 rpm - 50 Hz)		Engine speed (1800 rpm - 60 Hz)	
	PRP	ESP	PRP	ESP
Gross power (kW)	403	438	445	487
Spec fuel consumption. Full load (g/kWh)	186	192	195	199
Spec fuel consumption. 3/4 load (g/kWh)	183	183	187	188
Spec fuel consumption. 1/2 load (g/kWh)	185	184	190	187
Heat rejection to cooling water (kW)	119	134	142	166

The engine studied is from the Scania line for stationary engines and operates in two modes, PRP (Prime Power) for continuous operations where load variation occurs and ESP (Stand-by power) in which it is intended for operations under normal load, varying only when power failure occurs. For the analysis, the engine was used in the ESP operational mode due to the better initial assessment of the variables, as well as the load factor of the total engine, so that the maximum performance of the engine under study is evaluated.

### 2.2 Diesel RK software, fuels and computational modeling

Diesel RK software was developed between 1981 and 1982 at Bauman Moscow State Technical University in Moscow, Russia. It is a software designed to perform simulations on diesel engines presenting the thermodynamic variables related to the process of the engine under test. In addition, it is possible to optimize the performance of the engines under study. It presents precision in its results and can be used in engines of the Diesel and Otto cycle, where it is possible to apply a range of different fuels and compare their performance. It is a very intelligent system and simulates the variation of several parameters depending on several aspects, such as rotation.

The Diesel RK software has in its library a wide variety of fuels that can be implemented in the studies. Thus, for the study presented here, “Diesel N2” was used, which is one of the fossil fuels currently used and the biofuel “Biofuel SME” (B100). In addition to these two fuels, the simulations were carried out with their blends B20 (20% biofuel SME + 80% diesel N2) and B40 (40% biofuel SME + 60% diesel N2). It was chosen to work at the speeds of 1500 rpm and 1800 rpm for each fuel and the performance of the engine for the described configuration was analyzed. Table 2 shows the main physical and chemical characteristics of the fuels used.

The operating mode used in the engine was based on the concepts of application and optimization of the software itself, which are provided in the user guide of the software, which is illustrated in figure 1.

Table 2 – Main physical and chemical characteristics of the fuels used in the computational modeling

Fuel	Composition (mass fraction)				LHV (MJ/kg)	Cetane Number	Density at 323 K (kg/m <sup>3</sup> )
	C	H	O	S			
Diesel N2	0.870	0.126	0.004	0	42.50	48.0	830
B100	0.773	0.118	0.108	0.005	36.22	51.3	885
B40	0.829	0.123	0.047	0.00208	39.89	49.4	852
B20	0.849	0.124	0.025	0.001105	41.18	48.7	841

HP stage turbine settings		HP stage compressor settings									
#1	"RPM=1500, PR=4.69"	#6									
#2	"RPM=1800, PR= 4,335"	#7									
#3		#8									
#4		#9									
#5		#10									
Mode of Performance (#1 = Full Load)	<input checked="" type="checkbox"/> #1	<input checked="" type="checkbox"/> #2	<input type="checkbox"/> #3	<input type="checkbox"/> #4	<input type="checkbox"/> #5	<input type="checkbox"/> #6	<input type="checkbox"/> #7	<input type="checkbox"/> #8	<input type="checkbox"/> #9	<input type="checkbox"/> #10	
Engine Speed, [rpm]	1500	1800	1500	1500	1500	1500	1500	1500	1500	1500	
Cycle Fuel Mass, [g]	0,3433	0,3159	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	
Injection / Ignition Timing, [deg B.TDC]	8	8	12	12	12	12	12	12	12	12	
Ambient Pressure, [bar]	1	1	1	1	1	1	1	1	1	1	
Ambient Temperature, [K]	288,15	288,15	288	288	288	288	288	288	288	288	
Inlet Pressure Losses (before compressor), [bar]	0,02	0,02	0,02	0,02	0,02	0,02	0,02	0,02	0,02	0,02	
Differential Pressure in exhaust (tail) system, [bar]	0,04	0,04	0,04	0,04	0,04	0,04	0,04	0,04	0,04	0,04	
Compressor Pressure Ratio (HP Stage)	4,69	4,335	2	2	2	2	2	2	2	2	
Compressor Adiabatic Efficiency (HP Stage)	0,707	0,714	0,723	0,723	0,723	0,723	0,723	0,723	0,723	0,723	
Fraction of the Exhaust Gasflow By-passed before Turbine	0	0	0	0	0	0	0	0	0	0	
Fraction of the Airflow By-passed after Compressor into atmosphere	0	0	0	0	0	0	0	0	0	0	
Average Total Turbine Inlet Pressure (HP St) (or first appr.), [bar]	1,77	1,77	1,77	1,77	1,77	1,77	1,77	1,77	1,77	1,77	
Turbocharger Efficiency (HP Stage)	0,5	0,51	0,515	0,515	0,515	0,515	0,515	0,515	0,515	0,515	

Figure 1 - Operating mode of the Scania DC13 072A engine modeled on the Diesel RK software

Based on this operational mode, on the characteristics of the engine and on the properties of the fuels, computer simulations were performed for both engine speed using the fuels mentioned earlier.

### 2.3 Energy analysis

To carry out the energy analysis of the engine, some considerations were made: it was assumed that the engine operates under ideal conditions. The energy analysis is associated with the application of the first law of thermodynamics. Therefore, it is necessary to find the values of the heat supplied, the brake power and the heat lost by the cooling system as well as by the exhaust gases.

According to Panigrahi et al ,2014, the heat provided by the fuel is given by equation 1, which is the product between the mass flowrate used and the lower heating value of the fuel supplied.

$$\dot{Q}_f = \dot{m} * LHV \tag{1}$$

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The brake power ( $W$ ) can be written using the torque ( $T$ ) as (Sayin and Kahraman, 2016):

$$W (KW) = \frac{\omega * T}{60} \quad (2)$$

The energy lost in the exhaust gases ( $Q_{exh}$ ) is given by the difference between the heat supplied, the brake power and the heat rejection by the cooling system ( $Q_{cool}$ ). That is (Odibi et al., 2019):

$$Q_{exh} (KW) = \dot{m} * LHV + \frac{2 * \pi * T}{60} - Q_{cool} \quad (3)$$

An important point in this type of analysis is the amount of thermal energy the engine that is able to convert into brake power, which is called BTE (brake thermal efficiency) and is determined by the ratio of the brake power to the heat provided by the fuel. That is (Heywood, 1988):

$$BTE (\%) = \frac{W}{\dot{Q}_f} * 100 \quad (4)$$

Another relevant parameter is the BSFC (brake specific fuel consumption) which is a parameter that reflects the efficiency of the combustion process that occurs in an engine. Equation (5) shows the method to find the BSFC which is the ratio between the flow rate of mass of fuel by the brake power provided by the engine.

$$BSFC = \frac{\dot{m}_f}{W} * 100 \quad (5)$$

## 2.4 Exergy analysis

For the exergy analysis of the engine, the pressure of 101325 Pa and temperature of 288.15 K was assumed for the dead state. Equation 6 demonstrates the conservation of exergy in a control volume operating in steady state.

$$E_{cool} + E_W = \sum \dot{m}_{in} ex_{in} - \sum \dot{m}_{out} ex_{out} - E_{des} \quad (6)$$

Equation 6 shows that the exergy related to the cooling system ( $E_{cool}$ ) added to the brake power exergy ( $E_W$ ) is proportional to the difference of the exergy of the products ( $\sum \dot{m}_{out} ex_{out}$ ) and the exergy of the reactants ( $\sum \dot{m}_{in} ex_{in}$ ). Therefore, the amount of energy resulting from this equation is inherent to the irreversibility of the process and is called destroyed exergy ( $E_{des}$ )

The terms of the first part of the equation (6) are the transfer of exergy by heat and by brake power and are found from equations 7 and 8.

$$E_{cool} = \sum Q_{cool} \left(1 - \frac{T_0}{T_{exh}}\right) \quad (7)$$

$$E_W = \dot{W} \quad (8)$$

It is known that the exergy that enters the engine is related to the release of energy from the fuel. Thus, the first term of the second part of equation (6) is the product of the mass flowrate by the lower heating value of the fuel and a factor. That is (Panigrahi et al ,2014):

$$\sum \dot{m}_{in} e_{in} = \dot{m}_f LHV \phi \quad (9)$$

In which the factor  $\phi$  is found using equation (10). It is necessary to know the mass fraction of each element contained in the fuel (Sayin et al., 2007 apud Kotas, 1995).

$$\phi = 1,041 + 0,1728 \left(\frac{H}{C}\right) + 0,0432 \left(\frac{O}{C}\right) + 0,2169 \left(\frac{S}{C}\right) \left[1 - \left(2,0628 \frac{H}{C}\right)\right] \quad (10)$$

Where: H, C, O and S are the mass fraction of hydrogen, carbon, oxygen and sulfur, respectively.

The second term of the right side of equation (6) consists of the sum of the product between the mass flowrate and exergy of the products. In internal combustion engines, the exergy of the products is associated to the exergy of the exhaust gases.

The exergy efficiency ( $\eta_{II}$ ), or the Second Law efficiency, shows the ratio between the exergy recovered by the total exergy input on the engine (Sayin and Kahraman, 2016).

$$\eta_{II} = \frac{\text{recovered exergy}}{\text{Input exergy}} = 1 - \frac{\dot{E}_{des}}{\dot{m}_f LHV \phi} \quad (11)$$

### 3. RESULTS AND DISCUSSIONS

After completing all formulation and applying all the necessary values, the software Diesel RK could perform its modeling of the engine performance, allowing the achievement of the engine operational values. Through these values, it was possible to calculate the energy and exergy parameters of each type of fuel in each of the operating engine speeds. As it is a stationary engine, the speeds of 1500 and 1800 rpm were considered. Besides, an empirical study in other engine speeds to collect the operational conditions of the engine would be feasible.

#### 3.1 Energy analysis

In the energy analysis, it is possible to verify the energy supplied converted into other forms of energy during the internal combustion engine operational process. Therefore, it is possible to verify the percentage distribution of the input energy into other types of energy in the engine. Figure 2 shows this distribution for engine speed of 1500 rpm and 1800 rpm, considering Diesel N2 as the fuel.

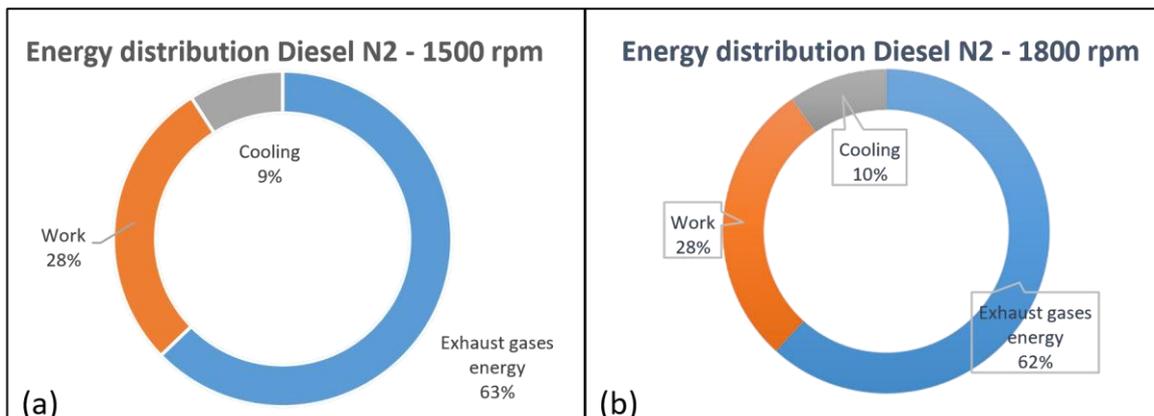


Figure 2 – Results of energy distribution for Diesel N2 as the fuel and engine speed of (a) 1500 rpm and (b) 1800 rpm

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As can be seen from Figure 2, the values of energy distribution for the engine speed of 1500 rpm compared to these values for the engine speed of 1800 rpm are presented without much variation for the work, and a variation of 1 percentage point for the cooling and the energy of the exhaust gases. Due to this small difference, the results for the other fuels will only be shown for the 1800 rpm engine speed. Thus, Figure 3 shows the results of energy distribution, for the engine speed of 1800 rpm, now for the fuels Diesel N2, B100 (100% Biofuel SME), B40 (40% biofuel SME + 60% diesel N2) and B20 (20% biofuel SME + 80% diesel N2).

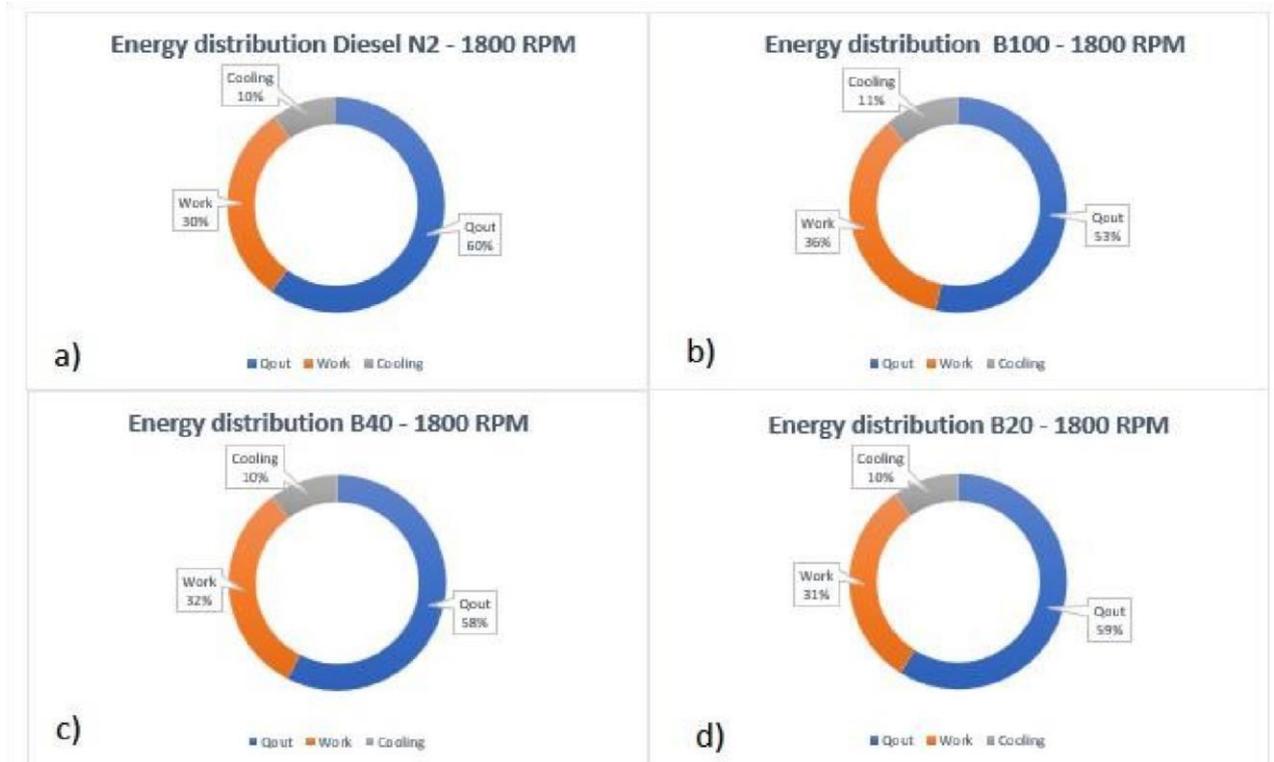


Figure 3 - Energy distribution of the engine operating with (a) Diesel N2, (b) 100% of biofuel SME, (c) B40 and (d) B20, at the engine speed of 1800 rpm.

It is possible to see through Figure 3 that the work output is better using B100 than Diesel N2. Their blends (B40 and B20) show close values. So, it is possible to notice that the B100 has a BTE value higher than all other fuels evaluated since they all present practically identical values of energy lost in cooling. The higher value in the cetane number shows that due to the high engine speed, the combustion of the B100 was more efficient than that of the other fuels, making the conversion into brake power more efficient.

Figure 4 shows the results of the energy distribution comparing the fuels used in the engine and Figure 5 show the Sankey diagram of each fuel used in the engine.

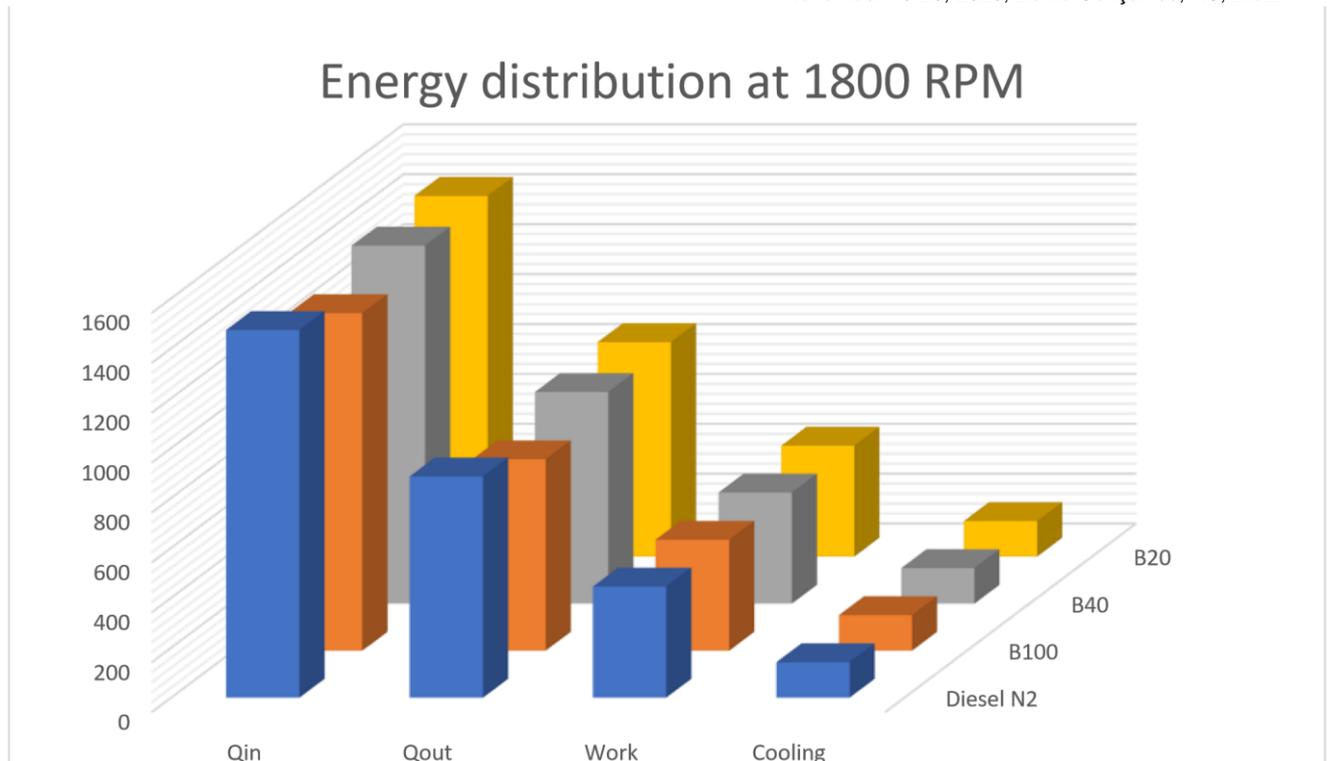


Figure 4 – Results of the energy distribution in the diesel engine comparing all fuels.

Even presenting a higher value for BTE, it is noticeable that the B100 performs less release of combustion energy, this fact is justified by the low LHV. Therefore, when presenting torque values approximate to the others fuels, its BTE value becomes higher than that of others. However, for such efficiency to occur, it is necessary to have a high mass flow of fuel of almost twice the amount of Diesel and a pressure and injection time higher than that of other fuels. These parameters most of the time present few conditions for modifications due to the relationships already predetermined in the injection system of the engines that are on the market.



Figure 5 – Sankey diagram of the internal combustion engine using (a) Diesel N2, (b) 100% biodiesel, (c) B40 and (d) B20

As observed in figure 5, the Sankey diagram in figure 6 shows the energy distribution that occurred in the application of each fuel in the engine. The diagram shows that the greater the energy available for a fuel to reach a certain power, it will be greater the loss of energy  $Q_{out}$ , therefore, for a better energy efficiency, you must insert the minimum energy necessary to obtain the necessary axis work, since cooling and brake power were considered constant for engine operation.  $Q_{out}$  energy is generated from various irreversibility of the process in question, such as pumping, friction, incomplete combustion, high temperature of the exhaust gases, among other variables.

Figure 6 shows the relation between BTE and BSFC for all fuels, the figure highlights the speed ranging from 1500 to 1800 RPM:

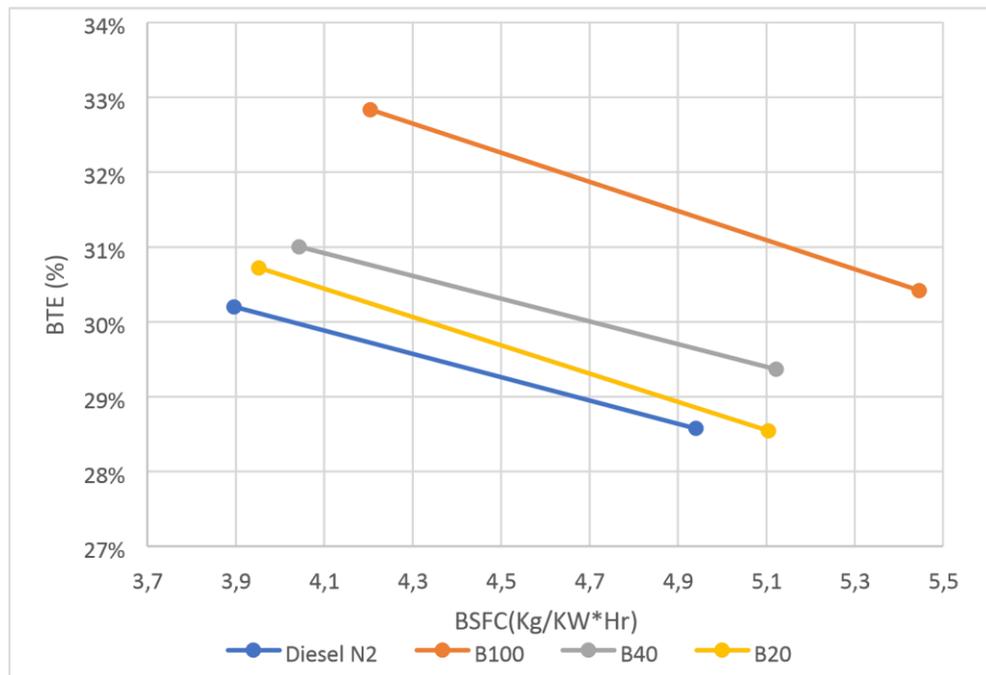


Figure 6 – Relation between BTE and BSFC

As noted, due to the better use of the energy released in combustion and brake power, the B100 presents better BTE values, followed by the fuels that present this in its composition. Another point already observed that is justified in figure 6 is the fuel consumption that at 1500 RPM has BSFC very close, however when reaching the rotation of 1800 RPM it is noticed that the B100 needs much more fuel to reach the power specifications of the motor.

### 3.2 Exergy analysis

In the exergetic analysis it is possible to observe that the values of the exergy present have a lower value than the energy plots, as an example the cooling system, according to the formulations previously demonstrated.

Qout energy was divided into two types for exergetic analysis, the exergy carried by the flue gases and the destroyed exergy which is the exergy caused due to the countless reversibilities that happen in thermal machines and which are the focal point for a better performance of the engines combustion, because by reducing these values better efficiency can be obtained.

Figures 7 and 8 show the relationship between first and second law efficiencies and BSFC with exergetic efficiency respectively.

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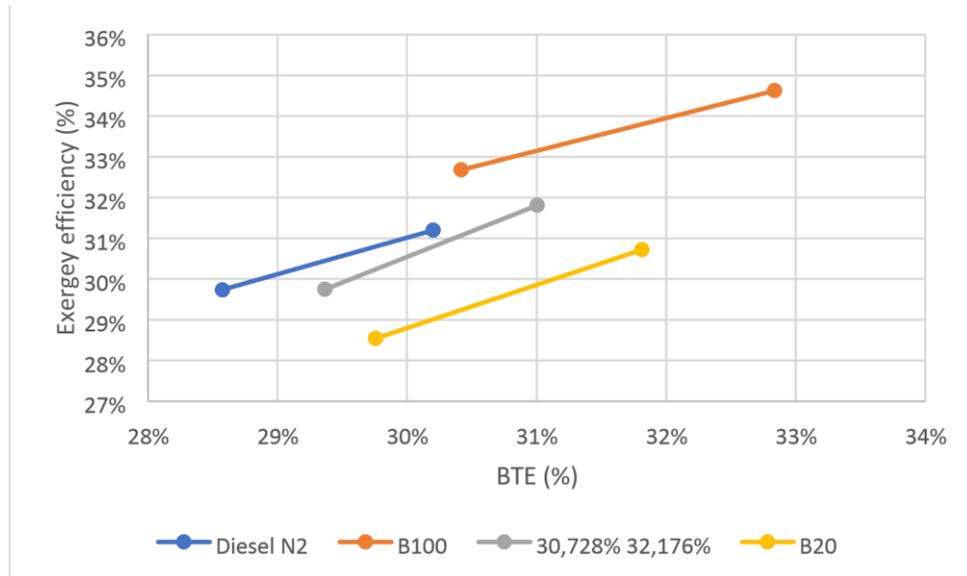


Figure 6 – Relation between BTE e exergy efficiency

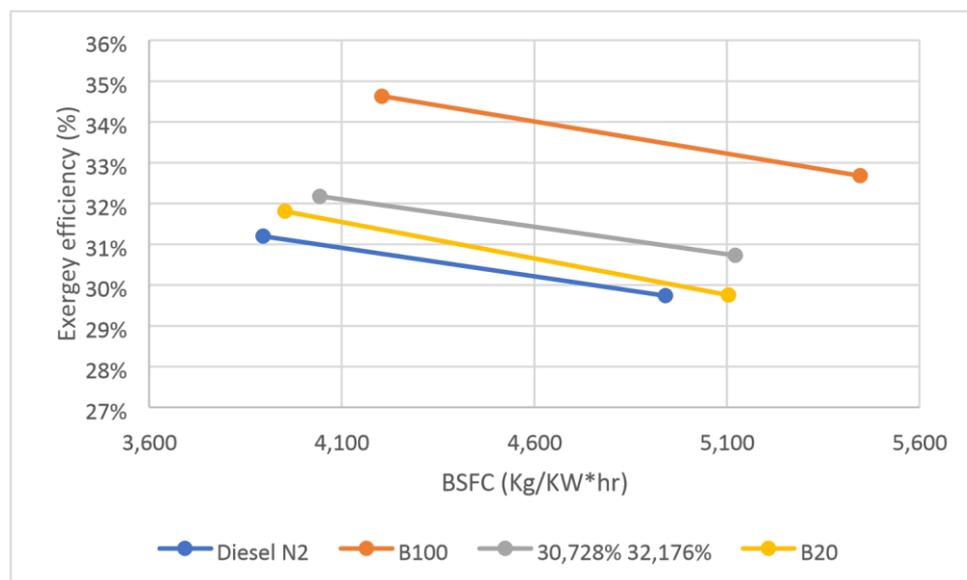


Figure 7 – Relation between BSFC e exergy efficiency

As can be seen, the B100 has major energy efficiency values than other fuels, because in this case, with the brake power and constant cooling, the high energy supplied by the fuel is lost as a destroyed exergy, since the exergy of the gases combustion for all fuels showed low values. Thus, there is a high fuel consumption and, consequently, a high fuel consumption, resulting in increased energy released and high exergetic efficiency.

As a quantity of mass greater than B100 greater than surplus is required to reach the required torque, greater release of chemical energy is obtained due to the greater fraction of chemical elements, however, the increasing of BSFC is certain viability to obtain a close exergetic performance or even higher than the other items sacrificed as a portion of the shaft work or use the RGE gas recirculation system, thus allowing to obtain the high energy value lost in the exhaust gases.

#### 4. CONCLUSION

This paper presents the energy and exergy analysis of an internal combustion engine using the Diesel RK software for different fuel used: Diesel N2, biofuel SME (B100) and their blends (B40 and B20). B40 is a blend containing 40% of biofuel SME and 60% of diesel N2 and B20 is a blend containing 20% of biofuel SME and 80% of diesel N2. The analyses intended to show the performance of alternative fuels in diesel engines through a simulation for a pre-feasibility analysis and acceptance of a project.

It is clear that the operational conditions contributed to some unexpected results. Therefore, in practical studies it is suggested to observe the physicochemical characteristics of biofuels for geometric and operational parameter of the engine to be set.

For the two engine speed (1500 and 1800 rpm), a “conservation” was observed in the percentage values in the energy balance, with little difference between the operational parameters. However, care must be taken when working with this since the difference in the form of energy occurs considerably when changes in speed at the same load takes place.

Parameters such as the lower heating value and the oxygen ratio directly interfere in the combustion and, consequently, in the performance of the engine. Another important variable is the cetane number, which is the reaction of the fuel at the combustion process beginning. This parameter showed higher number for the B100 fuel compared to the one for Diesel N2. This difference displayed a modified in the BTE values, that is, because the high RPM, the fuel energy was not efficiently converted into work due the time to realize the combustion.

In the exergetic analysis it is possible to verify a drop in efficiency due to the high mass of fuel needed to supply the same heat, as well as energies lost for fuel injection, this factor being somewhat harmful to the engines. But it is seen that taking the appropriate measures and optimizing the operation of the engine for the fuel in question, the B100 and blends becomes an excellent option as an alternative fuel, as it presents considerable energy values, physical and chemical properties close to that of Diesel and meets the demands energetic and with adjustments also meets the exegerical demands requested by the engines.

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